8. NOISE-ATTENUATING ELEMENTS

The noise-attenuating properties of pipes, ducts, tanks and vessels, in-line and vent silencers, lagging and propagation outdoors and in rooms is addressed below.

Pipes, ducts, tanks and vessels within which the gas flows act to constrain the sound field through mass and stiffness. Too much acoustic energy within the pipe can have negative consequences however. "Excessive vibration can cause failure or damage to valve and pipe mounted instruments and accessories. Piping cracks, loose flange bolts, and other problems can develop".[42]

The open end of a pipe has a noise attenuating function: some of the sound energy is reflected back from the opening when the wavelength is significantly larger than the pipe exit, the exit is rather abrupt, and high velocity discharge flow is not present.

In-line silencers, if properly selected, are an effective means of reducing noise levels. They accomplish this task by converting acoustic energy into minute amounts of heat energy. Most practical silencers cause a measurable pressure drop. Generally speaking, the more sound that can be attenuated per unit distance, the more pressure drop the silencer develops. One consequence of the pressure drop is flow noise generated within the silencer itself, called *self-noise.* Thus some judgement must be exercised in balancing the competing interests of attenuation and pressure drop.

Lagging of ducts, pipes and vessels attenuate sound as it radiates from pipe and duct walls. The lagging constrains the sound in a sound-absorbing cavity that converts the sound into minute amounts of heat.

8.1. Pipes and Ducts

The noise attenuating performance of pipes and ducts is called in *Transmission Loss*, which is a measure of the ability of the wall to resist transmission of sound. High values of transmission loss correspond to a high degree of sound isolation.

8.1.1. Transmission Loss of Circular Pipes

The Transmission Loss of a circular pipe[43] reaches a minimum at the first mode cut-on frequency of the pipe:

$$TL_{f_0} = 10\log\left[\frac{rt_p^{\ 2}}{D_p^{\ 3}}\left(\frac{P_2}{P_a}+1\right)\right] + 69.5 + \Delta TL(f, f_0, f_r)\, dB$$

The first mode cut-on frequency f_0 and the ring frequency of the pipe wall f_r are:

$$f_0 = \frac{0.586c}{D_p}$$

$$f_r = \frac{C_L}{\pi D_p}$$

The correction term ΔTL is positive-valued, such that Transmission Loss increases as the frequency moves away from the lowest value at f_0. Strong low frequency attenuation comes about because the pipe walls must be literally stretched by hoop stress in order for the pipe walls to vibrate uniformly (n=0 mode). At f_0, the sound field within the pipe is no longer uniform across the cross-section, allowing other more efficiently radiating pipe wall vibration modes to become active. Above f_r, the radius of curvature of the pipe wall is large compared to a wavelength, and the wall behaves like a flat plate. The Transmission Loss of flat plates increases with frequency.

$$\Delta TL = -20\log\frac{f}{f_0} \text{ for } f \le f_0$$

$$\Delta TL = 13\log\frac{f}{f_0} \text{ for } \mathrm{f}_0 < f \le f_r$$

$$\Delta TL = 20\log\frac{f}{f_0} - 7\log\frac{f_r}{f_0} \text{ for } f > f_r$$

For steel pipe, c_L is 5050 m/sec (16,564 ft/sec) and f_r is $c_L/\pi D_p$.

8.1.2. Transmission Loss of Rectangular Ductwork

The Transmission Loss of rectangular ductwork[43] is equal to that for circular pipe at high frequency, but is typically less at low frequency because of the reduced bending stiffness of the walls. Especially where control of low frequency noise is important, consideration should be given to using circular pipe.

Transmission Loss performance of a duct wall (assuming single as opposed to double layer construction) follows a "mass law", in which mass per unit area and frequency are the only relevant parameters:

$$TL_{duct} = 20\log(f\rho_s) - 45 \text{ (dB)}, f \ge f_{cr}$$

$$TL_{duct} = 13\log\left(\frac{f\rho_s^{\ 2}}{a+b}\right) - 13 \text{ (dB)}, f < f_{cr}$$

$$f_{cr} = \frac{0.520c}{\sqrt{ab}}$$

where ρ_S is mass per unit area in kilogram per square meter and a and b are duct cross-sectional dimensions in meters.

8.1.3. Structural Acoustical Limits

Valve manufacturers recommend limiting control valve noise to 115-120 dB(A) at 1 meter[42,44]. Given that most circular pipe in which control valves are installed has Transmission Loss on the order of 50 dB, the corresponding interior sound pressure levels is on the order of 165 to 170 dB(A). Indeed, one study[45] indicates that the maximum allowable sound power level to avoid structural failure for pipe with 8 mm wall thickness varies from 170 dB for 10-in. diameter pipe to 160 dB for 36-in. diameter pipe. The function has been parameterized for the purposes of this study as

$$PWL_{Limit} = 185.5 - 1.5\left(\frac{D_p}{1\,\text{in.}}\right) + 0.02\left(\frac{D_p}{1\,\text{in.}}\right)^2$$

$$\text{for } 10\,\text{in.} \le D_p \le 36\,\text{in.}$$

Noise control should be implemented at the source when sound power levels exceeding the structural limit are encountered. Exterior lagging and other "add on" noise control treatments that do not reduce the interior noise level or pipe wall vibration are ineffective.

Note – the structural fatigue criterion given above was developed for petrochemical plants and refineries where continuous operation is usually assumed. In cases of infrequent operation the criterion could probably be relaxed somewhat. The criterion could also probably be relaxed somewhat for pipes with thicker walls. No data is available for either case at this time.

8.2. Acoustical lagging

Acoustical lagging refers to the treatment of piping and equipment to reduce the radiation of noise to surrounding areas. Pipe lagging performance is expressed in terms of *Insertion Loss*; high values indicate a high degree of acoustical isolation.

The sound pressure level L_P after installation may be computed from that before installation as:

$$L_{P,after} = L_{P,before} - IL$$

Lagging is selectable by thickness on the System spreadsheet and in the Flow Noise spreadsheet.

Lagging consists of a layer of flexible, high-density sound absorbing material applied directly to the exterior surface of the pipe. A massive, continuous jacket layer is applied over the absorbing material. The jacket constrains sound within the acoustic cavity where some of the sound energy is converted into heat energy.

At low frequencies the entrapped air in the cavity is stiff (with stiffness proportional to the inverse of the cavity depth) and provides an unattenuated path for vibration to travel directly to the jacket, bypassing the acoustical insulation. The jacket adds very little mass to the system, hence negligible attenuation is achieved under these circumstances. Also, it should be observed that to extend low frequency performance, the cavity depth must be increased.

At high frequencies the air in the cavity is less stiff and sound must travel through the acoustical insulation, which attenuates the wave as it travels, until it reaches the jacket, which reflects it back into the cavity. Significant levels of attenuation are achievable provided that

- the jacket is continuous, and
- no significant rigid paths (such as supports) have been introduced between the pipe wall and the jacket.

The absorbing material is typically glass fiber (2½ to 6 pounds per cubic foot density) or mineral fiber (4 to 8 pounds per cubic foot density). Other materials such as calcium silicate and expanded closed-cell foams are not recommended because they are too rigid. When calcium silicate or closed cell foam are desired for thermal isolation, a thin layer should be used next to the pipe. The acoustical lagging provides good thermal insulation as well.

The jacket material is typically 26 to 28 ga. Steel, 16 to 20 ga. Aluminum, or a barium-loaded vinyl material. A common factor among these is that the surface density is approximately 1.25 pounds per square foot. Lead/aluminum laminate has been used in the past.

If periodic inspection is required, a lace-up style removable/reusable blanket may replace the jacket and perhaps the blanket as well. Note that a removable/reusable blanket is susceptible to degradation with wear and the possibility of gaps developing during re-installation.

8.2.1. Lagging Specification

The acoustical lagging shall consist of 2-in., 4-in., or 6-in. thick mineral fiber placed against the pipe wall plus an external jacket incorporating steel, aluminum and/or loaded vinyl to achieve a 1.25 pound per square foot surface weight. If loaded vinyl is used, it shall be sheathed with an exterior metal jacket. Thermal insulation such as calcium silicate or closed-cell synthetic foams shall not be acceptable substitutes for the cavity fill.

All circumferential joints of the insulation should be staggered and sealed with a non-hardening adhesive. Longitudinal seams and adjoining sections are to be firmly butted together and sealed. All gaps and voids are to be packed with loose insulation. Field cut the acoustic insulation to snugly fit around irregular shapes, elbows, flanges and valves. Jacket seams shall overlap by no less than 2 inches; stainless steel banding shall be applied on 9-10 inch centers (use of screws and rivets alone is not recommended).

Insertion Loss performance of the lagging system shall be no less than given below in Table 13 when measured in accordance with ASTM E1222 "The Laboratory Measurement of the Insertion Loss of Pipe Lagging Systems" or by a field test method acceptable to the purchaser.

Table 13: Insertion Loss Performance of Lagging Systems

	31.5	63	125	250	500	1000	2000	4000	8000
2 in.	1	3	4	6	12	22	23	21	20
4 in.	2	4	5	10	15	27	30	24	20
6 in.	4	7	10	15	25	30	30	22	20

8.3. Radiation and Reflection from the Open End of a Pipe

The process of radiation and reflection of sound from the open end of a pipe is well understood in the absence of mean flow. Reflection of sound is most pronounced when the pipe termination is abrupt (rather than extended by means of a horn). A short bell-mouth is considered abrupt for the purposes of this Guide.

Sound having wavelength greater than the pipe opening diameter reflects back into the pipe; sound having wavelength less than the pipe opening diameter propagates freely out into the environment.

The introduction of mean flow complicates the matter considerably. Consider first the limiting cases. For discharge flow $M_j \geq 1$ "reflected" sound is unable to travel upstream into the pipe, and is convected out into the environment with the flow. Thus, no reflection loss occurs in this case: all of the gas-borne sound in the pipe is radiated. Conversely, for intake flow with $M_j < -1$, no sound within the pipe can reach the plane of the inlet: it is convected back into the pipe with the flow. The reflection loss in the latter case is complete: no sound can be radiated.

In reality, however, sound is generated by a high velocity inlet vent. The most probable sources are inlet debris screens, sharp edges near the opening where the mean flow velocity is not yet sonic, and constrained jet noise downstream of the inlet radiating out through the pipe walls.

The following parametric dependence for the reflection loss (IL) has been deduced from data given in two theoretical and empirical studies[46,47]:

$$IL = (1+M)^2 (1 - r_E) \left(\frac{\rho_a c_a}{\rho_1 c_1} \right)$$

where

$$r_E = e^{-\xi_1(ka) \cdot ka} \left(1 - e^{-\xi_2(M)\sqrt{ka}} \right)$$

$$\xi_1(ka) = 0.583 + 0.391(ka)^2$$

$$\xi_2(M) = \left(\frac{1-M}{7.17M^2 + 0.43M} \right)$$

where IL denotes the Insertion Loss or reduction in sound power level due to the effect, M is positive for discharge and negative for inlet flows, a is the radius of the pipe opening and k is the acoustic wavenumber $2\pi f/c$. The index 0 and 1 for the density and sonic velocity factors refer to ambient and within the pipe, respectively.

8.4. Silencers

Four basic types of in-line silencers exist: dissipative silencers, reactive silencers, combination silencers and vent silencers. With the exception of the vent silencers, all of these types may be used for in-line service. For the purposes of this Guide, silencers are assigned the generic descriptors D, R, C and V, respectively. Most silencers come in various diameters and sizes to accommodate a variety of flows and performance ranges. Four generic grades of performance are referred to in manufacturers' literature: commercial, semi-residential, residential and critical. These grades refer to increasing degrees of performance associated with the named applications, and are assigned generic descriptors –2, -3, -4, and -5. More detailed information on silencers is available from Universal[48] and Burgess-Manning[49].

Silencer performance is expressed in terms of *DIL (Dynamic Insertion Loss)* which is the Insertion Loss under actual service conditions of flow, temperature, etc.

Actual silencer DIL performance is strongly affected by a number of parameters. Silencer performance figures tabulated below are generic and are for preliminary design purposes only. Silencer performance figures for the actual service conditions anticipated should be requested from silencer manufacturers.

For silencer conditions exceeding 15 psig pressure and 20 in. Hg vacuum, ASME Code construction (Section VIII, Div. 1) is typically recommended. It should be noted that higher temperatures alter the effective properties of the acoustic fill in dissipative silencers and require larger volumes for reactive silencers. Acoustical absorption materials are typically rated for temperatures not exceeding 325 °F, while the silencer bodies are typically rated for 500 °F. Make sure that the absorbing fill is rated for the entire range of expected flow temperatures.

The in-line silencers typically associated with control valves have special design considerations and are not addressed here.

8.4.1. Dissipative Silencers

Dissipative silencers attenuate sound by placing sound absorptive materials in contact with the flow. They tend to perform better at higher frequencies. Increased length, greater depth of fill, and narrow flow passages contribute to improved acoustical performance. The flow resistance of the acoustical fill must be carefully controlled to ensure optimum performance. Their performance can be degraded by the presence of oil, dust or other contaminants. Fill erosion can also be a problem above 6000 feet per minute. In such cases, fill protection can be improved, but at the expense of high frequency performance. High performance dissipative silencers have a pressure drop approximately equal to one velocity pressure head ($K=1$).

Dissipative silencers are ideal for axial compressor inlets, fans and blowers, some very low pressure vents (< 15 psig) and other applications where primarily high frequency sound (above 500 Hz) is to be attenuated and low pressure drop is required.

Dissipative silencers are constructed in several configurations. Generic designations have been assigned to the silencer types to facilitate incorporation into the workbook.

- Concentric (DC): sound-absorbing material in a recessed cavity behind perforated metal. The flow path is straight with no restrictions. This type of silencer produces very little pressure drop, but must be many inlet duct diameters long to achieve moderate levels of performance. Two silencer types are documented: DC-2 and DC-4, which refer to commercial and residential grade concentric dissipative silencers.
- Annular (DA): sound-absorbing material is located behind perforated walls and within a streamlined, sound-absorbing centerbody. The flow path is altered by the presence of the centerbody, hence pressure drop is greater than for a concentric silencer. Higher DIL performance is possible in a more compact package. Three annular types are documented: DA-3, DA-4 and DA-5, corresponding to typical semi-residential, residential and critical grade silencers.
- Splitter (DS): sound-absorbing material is located in streamlined, parallel baffles placed in the flow. Performance is controlled by the percent open area (POA), the splitter depth and the ratio of length to splitter gap width. DS-25, DS-33 and DS-50 correspond to splitter silencers with 25%, 33% and 50% open area respectively.
- Tubular (DT): sound-absorbing material is packed into a volume that is traversed by a number of perforated parallel tubes that carry the flow. Similar DIL performance as a splitter silencer can be obtained in about 2/3 the length, but with increased pressure drop. DT-33-1, –2 and –3 refer to three lengths of this type of silencer.

Typical dissipative silencer performance is tabulated below in Table 14, along with the pressure loss factor *K*, typical length to silencer diameter (*L/D*) and length to inlet pipe diameter (*L/P*) ratios, and typical percent open area figures.

Silencer Pressure Drop can be estimated from

$$\Delta P = K \times \tfrac{1}{2} \rho U^2$$

Table 14: Typical Dissipative Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
DC-2	3	4	7	14	20	20	16	10	0.25	4.5	6.4	95
DC-4	5	10	20	30	40	45	40	35	0.25	5.0	13.0	95
DA-3	5	7	11	22	32	32	28	22	0.85	2.0	2.3	50
DA-4	5	8	14	24	34	36	32	26	0.85	2.8	3.7	50
DA-5	5	11	20	30	40	43	40	35	0.75	2.2	4.4	50
DS-50	10	22	30	35	38	34	23	15	0.60	4.0	4.0	50
DS-33	10	25	35	45	50	50	45	35	0.70	4.0	4.0	33
DS-25	10	26	40	55	60	63	60	50	0.90	4.0	4.0	25
DT-33-1	7	9	12	17	21	22	20	17	0.80	1.0	1.0	33
DT-33-2	10	16	22	33	42	44	41	36	0.90	2.0	2.0	33
DT-33-3	12	20	30	45	58	60	57	50	1.00	3.0	3.0	33

Dissipative silencer DIL decreases with increasing discharge velocity (where the sound travels with the flow). Conversely, the DIL increases on an intake system.

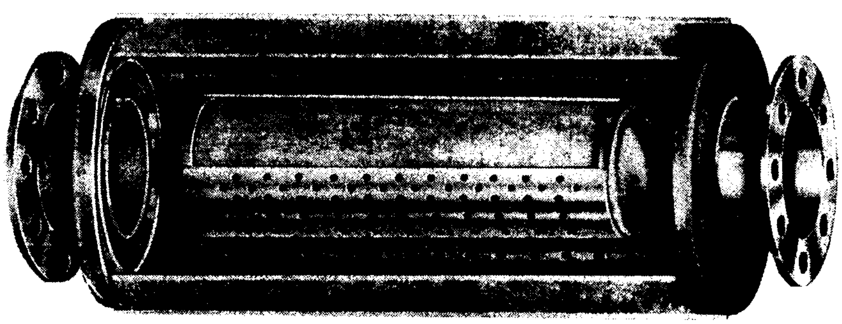

Figure 11: Dissipative Silencer

(Burgess-Manning[49])

8.4.2. Reactive Silencers

Reactive silencers attenuate sound by presenting an acoustical impedance that reduces passage of the acoustic wave. This is accomplished by using one or more chambers connected by tubes. They tend to perform better at low frequencies. Increased volume and number of chambers contribute to improved acoustical performance. High performance reactive silencers have a pressure drop approximately equal to four velocity pressure heads ($K = 4$). Lower pressure drop configurations are available, but performance is reduced.

Reactive silencers are appropriate for rotary lobe and reciprocating compressors, and any application where low frequency noise is to be attenuated and significant pressure drop can be tolerated.

Reactive silencers tabulated below cover low (L) and high (H) pressure drop ranges and all four generic grades of performance –2 through –5.

Table 15: Typical Reactive Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
R-2-L	10	20	28	22	15	13	10	8	0.5	3.0	7.1	50
R-2-H	12	20	27	23	18	17	16	15	4.2	3.0	7.0	50
R-3-L	16	28	35	28	20	15	12	10	1.0	3.7	9.8	50
R-3-H	16	25	33	27	23	20	20	20	4.6	3.4	8.0	50
R-4-H	20	30	35	30	27	25	24	24	5.0	3.7	9.8	50
R-5-H	25	35	36	35	32	29	28	28	5.3	4.4	11.8	50

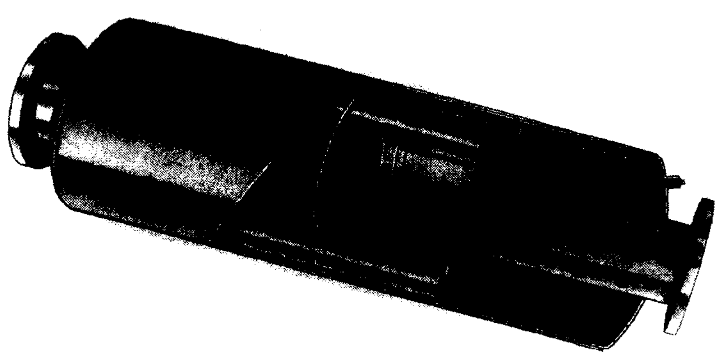

Figure 12: Reactive Silencer

(Universal Silencer[48])

8.4.3. Combination Silencers

Combination silencers attenuate sound using one or more elements of each of the dissipative and reactive type to achieve an insertion loss spectrum combining the benefits of both types. Combination silencers are often used on rotary lobe blowers and compressors.

Combinations tabulated below include: DCR, a short dissipative concentric silencer followed by a reactive chamber, VDR, a diffuser basket followed by a lined reactive chamber, VDA, a diffuser basket followed by a simple dissipative annular silencer, and three grades of VDC, a diffuser basket followed by a dissipative concentric silencer.

Table 16: Typical Combination Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
DCR	12	20	30	35	35	22	20	15	1	2.60	6.00	50
VDR	21	25	29	35	38	38	37	34	13	4.80	17.00	50
VDA	12	21	23	25	34	42	44	43	10	2.25	7.50	50
VDC-3	15	22	30	36	39	38	35	25	10	5.30	13.25	50
VDC-4	19	28	38	43	44	48	57	50	20	7.00	17.50	50
VDC-5	20	40	53	55	53	59	65	61	30	8.60	21.50	50

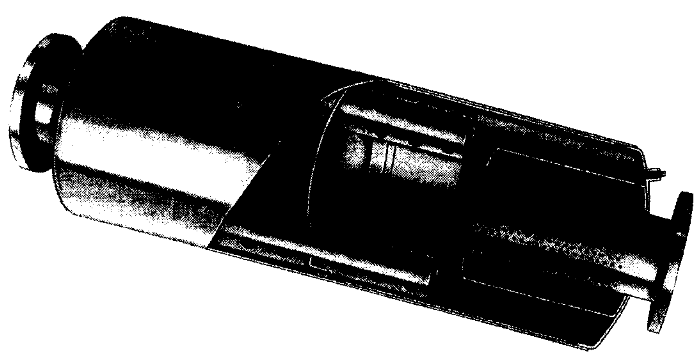

Figure 13: Combination Silencer

(Universal Silencer[48])

8.4.4. Vent Silencers

Vent silencers are a special type of dissipative silencer used to reduce noise from high velocity gas discharges. The vent silencer consists of one or more diffuser baskets that break the discharge jet into a number of smaller jets. This has the effect of shifting the peak frequency f_p upward by several octaves. A dissipative splitter silencer follows the basket. With the peak frequency shifted, the splitter silencer can achieve high *DIL* performance in a short distance. Vent silencers have a pressure drop approximately equal to ten velocity pressure heads per diffuser basket.

Note that sounds already present in the gas flow before reaching the outlet are not frequency-shifted by the diffuser basket. Some of the sound is reflected back into the pipe. In any event, attenuation of low-frequency sound energy in the flow should not be expected to be as dramatic as for the jet mixing noise: in this case, the vent silencer functions as a simple dissipative silencer.

The designation 2VS used below refers to two diffuser baskets applied in series. Four grades each of type VS and 2VS are documented.

Dynamic Insertion Loss performance of silencers for gas vent applications is tabulated below:

Table 17: Typical Vent Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
VS-2	7	6	9	14	19	21	20	19	11.75	2.0	5.1	50
VS-3	10	11	16	25	31	33	32	30	12.00	2.7	6.8	50
VS-4	13	17	24	36	44	46	43	40	12.25	3.4	8.5	50
VS-5	17	22	32	47	56	58	56	50	12.50	4.1	10.2	50
2VS-2	12	10	13	17	21	23	22	21	20.60	2.0	5.1	50
2VS-3	15	15	20	28	33	35	34	32	21.00	2.7	6.8	50
2VS-4	18	21	28	39	46	48	45	42	21.40	3.4	8.5	50
2VS-5	22	26	36	50	58	60	58	52	21.90	4.1	10.2	50

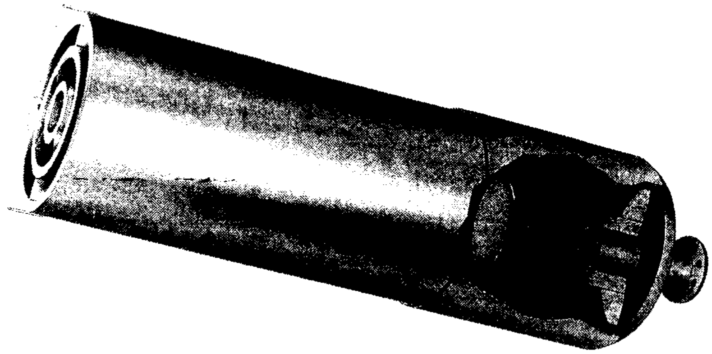

Figure 14: Vent Silencer

(Universal Silencer[48])

8.4.5. Silencer Self-Noise

The term self-noise in relation to a silencer refers to noise generated by the flow of air through the silencer. If poorly selected, the flow noise can severely impact the net performance of the silencer. The silencer self-noise in octave bands can be estimated (after Beranek and Ver[51]) as:

$$L_W = -145 + 55\log_{10}\left(\frac{U}{\text{ft/min}}\right) + 10\log_{10}\left(\frac{A_F}{\text{ft}^2}\right) - 45\log_{10}\left(\frac{POA}{100}\right) - 25\log_{10}\left(\frac{460 + T(°F)}{530(°F)}\right)$$

An extra term has been added to L_W to account for gases other than air:

$$L_W' = L_W + 10\log_{10}\left(\frac{MW}{28.967}\right)$$

This sound power level is added to the sound power level leaving the silencer on the quieter side.

In Vér's analysis, the self-noise is presented as constant across all octave bands. To account for the experience of others[48] the following *ad hoc* corrections are recommended:

Table 18: Proposed Octave Band Corrections for Silencer Self-Noise

	31.5	63	125	250	500	1000	2000	4000	8000
L_W corrections	+15	+10	+5	0	0	0	0	0	0

8.5. Sound In Enclosed Spaces

In the absence of reflecting obstacles sound waves decay as the travel. The reduction in level over distance is equal to $20 \log_{10}(r_2/r_1)$, where r_2 and r_1 are distances from the noise source. This type of propagation is often referred to as *direct sound* and corresponds to 6 dB reduction per doubling of distance.

Proximity to noise sensitive areas should be considered when siting noisy equipment. For instance, it is theoretically possible to achieve a 10 dB reduction by increasing the distance between equipment and observer from 10 ft. to 30 ft. Further reductions of this magnitude, however, are more difficult because of the practical distance scales involved within buildings, between laboratory buildings, and between the laboratory and the community.

The Sound Pressure Level recorded at a particular location is a function of:

- the sound power level (L_W) and directionality of the source, and
- the reflective and absorptive properties of the environment.

Sound seldom travels in an environment without reflecting obstacles. Within buildings, the radiated Sound Power is reflected by the room surfaces and the reflected sound energy trapped within the room distributes itself more or less uniformly as *reverberant sound.* The total sound pressure level at a point is therefore the sum of the reverberant sound pressure level and the direct sound pressure level.

The net numerical difference between the sound power level and Sound Pressure Level is often referred to as the *wave divergence.* A full treatment of sound in rooms is available in Beranek[50], Beranek and Vér[51], and NASA[52].

It should be noted that outdoor spaces can also be reverberant. Courtyards and other areas between buildings provide reflecting surfaces that cause sound pressure level to increase locally. Sky and other open areas provide sound absorption. The ground, even when landscaped, should be considered reflective.

From a standpoint of controlling noise in rooms, there are two options that should be pursued in the following order:

- Identify the distance r between the equipment and the noise sensitive area.
- Determine the distance $r_{eq}(f)$ at which the direct and reverberant sound levels are approximately equal.
- Compute the effect on the sound pressure level, either in octave bands or A-weighted as expressed by the criterion.
- Add sound absorbing materials to the room to increase the room constant R until r_{eq} is greater than or equal to r. Once this point has been reached, only incremental gains can be obtained by adding more sound absorption.

- Further reductions must come from source noise control or by means of noise control barriers or enclosures.

8.5.1. Room Acoustics as Implemented in the *Specifications Guide*

In the *Specifications Guide*[53], room acoustics is treated in an extremely general way. Surfaces and surface treatments are identified as either sound-absorbing or sound-reflecting by comparison to a list. The total area of sound-absorbing surfaces S_A is compared to the total area of sound-reflecting surfaces S_R. A reverberant condition is deemed to exist if

$$S_A > 160\left(1 - \frac{S_R}{2000\,\mathrm{m}^2}\right)\mathrm{m}^2$$

For the purposes of the *Specifications Guide*, a 5 dB(A) sound level increase is assumed to result from the reverberant condition. The non-reverberant condition is assumed to produce a 0 dB(A) sound level increase.

8.5.2. Room Acoustics Equations in *Design Guide*

For the purposes of the *Design Guide*, the classical room acoustics equation has been adapted for use:

$$L_{P,total} = L_W + 10\log_{10}\left(\frac{D(\theta)}{4\pi r^2} + \frac{4}{R(f)}\right)$$

$$R(f) = \frac{(S_A + S_R)\bar{\alpha}(f)}{1 - \bar{\alpha}(f)}$$

$$\bar{\alpha}(f) = \frac{S_A}{S_A + S_R}\alpha(f)$$

The first term in parentheses corresponds to geometric spreading of the direct sound, the second term corresponds to geometric spreading of the reverberant sound. This equation is applied in each octave-band because of the frequency-dependent performance of sound absorbing materials. When $R(f)$ is large because of large surface area or high degree of sound absorption, the direct field tends to dominate. Conversely when $R(f)$ is small, as it would be in small rooms or those with little sound absorption, the reverberant sound tends to dominate and the Sound Pressure Level reaches a constant value at some distance from the equipment. Note that because sound absorbing materials are more efficient at high frequency than at low frequency, it is possible for the direct field to dominate at high frequencies and the reverberant sound field to dominate at low frequencies.

In a formal engineering project, estimates of the sound absorption coefficients of all surfaces are collected to estimate *R(f)*. To simplify the work somewhat for the *Design Guide* Workbook, the estimated percent coverage of all room surfaces by sound-absorbing materials is used (the percentage includes the floor). All sound-absorbing surfaces are assumed to have the generic sound absorption given in Table 20 below. All non-absorbing surfaces are assumed to have uniform sound absorption of 5% at all frequencies.

Table 19: Sound-Absorbing and Sound-Reflecting Materials

Sound-Absorbing Materials	Sound-Reflecting Materials
Glass Fiber, 50 mm or thicker	Brick, Stone, Concrete
Mineral Fiber, 50 mm or thicker	Wood, Glass, Metal
Basalt Wool, 50 mm or thicker	Tile, Plaster
Open-Cell Foams, 75 mm or thicker	Gypsum Board
Tectum on 40 mm airspace, 50 mm or thicker	Closed-Cell Foams
Acoustical Ceiling Tile, on 400 mm airspace	Ground
Hanging Acoustical Baffles, 50 mm or thicker	
Sky, Open Doors, Open Windows, "Missing" Walls	

Table 20: Generic Sound-Absorption Coefficients for Sound-Absorbing Materials

	31.5	63	125	250	500	1000	2000	4000	8000
$\alpha(f)$	0.05	0.10	0.20	0.40	0.70	0.90	0.90	0.90	0.90

8.5.3. Direct Field for Large Equipment

The direct sound portion of the classical L_P equation is derived on the assumption that the source is a point source, which condition is satisfied if the distance r to the source is large compared to the characteristic dimension of the source. Because the sound power is assumed to be concentrated at a point, extremely high sound pressure levels are predicted for small values of r. This is not the case for large equipment when the sound power is distributed across the surface, such as would occur with radiation from a large pipe. Sound level increases at nearby locations are far less dramatic than the equation would indicate.

To take the geometry of the source into account in the equation, we recommend substituting the following effective distance r' for the radius r in room acoustics equations.

$$r' = \left(\sqrt{r^2 + \left(\frac{H}{2}\right)^2} \sqrt{r^2 + \left(\frac{W}{2}\right)^2} \right)^{\frac{1}{2}}$$

where H and W are the height and width of the source as viewed from the observation point. This equation is effective for extended surfaces and for line sources such as pipes (where H would be small).

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